ENGR7003:

NVH Coursework-2

Acoustic analysis of a car cavity.

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Introduction

Noise, Vibration and Harshness are among the crucial aspects that define a ride's quality and comfort. Modern car's interior sound quality is one of the most crucial customer requirements and is also a key characteristic from a sales point of view. The fine-tuning of the vehicle cabin for noise, vibration, and harshness has become a major challenge for OEMs, because of the light weighted structures. The internal noise level is greatly influenced by different sound sources like engine, motor, exhaust system, tyres, wind noise etc. The noise generated inside the cabin of the vehicle can be categorised as airborne or structure-borne. Therefore, it is important to understand the root cause of the problem and the impact on various bodies of the vehicle to minimise the external factors and isolate the cabin to meet the comfort and quality requirements. Presently, studies are being conducted in this domain.

The main objective of the following coursework is to perform an acoustic modal and harmonic analysis of a passenger car cabin to study and understand the influence of external factors on the sound level within the car in addition to acoustic absorption.

The Audi Q2 is the vehicle model chosen for the analysis in the following assignment.

Methodology

The first step in this study involved 3D modelling of the vehicle cabin in SOLIDWORKS using the vehicle's reference dimensions obtained from the internet. As this study is focused on understanding acoustic behaviour inside the vehicle, only the cavity/passenger cabin was considered to simplify and reduce the complexity. The first step in the FE method, modal acoustics analysis was performed to determine the structure's dynamic characteristics by identifying the acoustic structure's resonance frequency and modes. Further, a series of harmonic analysis were performed to measure the structure's acoustic response to external forces with and without the absorbing elements.

The convergence study for the model was performed to optimize the mesh element size required for the following analysis. The tetrahedral mesh type was used since it is best suitable for complex 3D geometry models as higher number of nodes results in better accuracy. The range taken into consideration for the convergence study was (0.02m - 0.2m). Based on the study, 0.05m was chosen as the optimal mesh element size for the analysis because there was not much deviation in the results when compared to the smallest element size, also it was efficient in terms of time and processing resources.

The below graph describes the results of the convergence study.

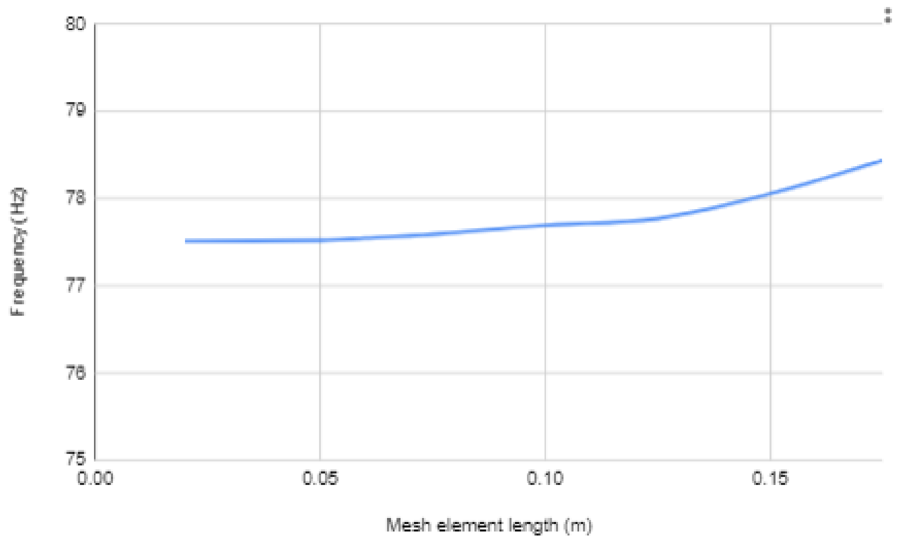

Figure 1 Convergence Study

Discussion on mode shapes

The modes describe characteristics of the structure which depends on the properties of the material and boundary conditions, these modes aid to predict the dynamic behaviour of the structure in operating condition. Since the exterior surface of the cabin was treated as a rigid wall during the analysis, no particular boundary condition was required.

The first three mode shapes are discussed below.

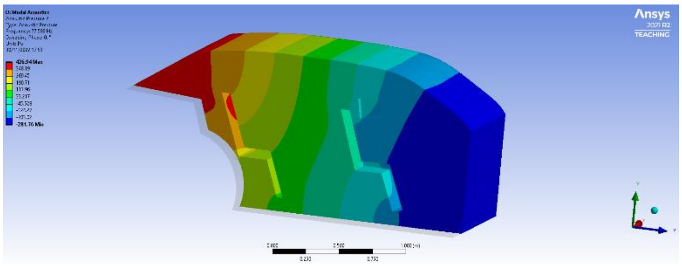

Figure 2 Mode 1 (77.588Hz)

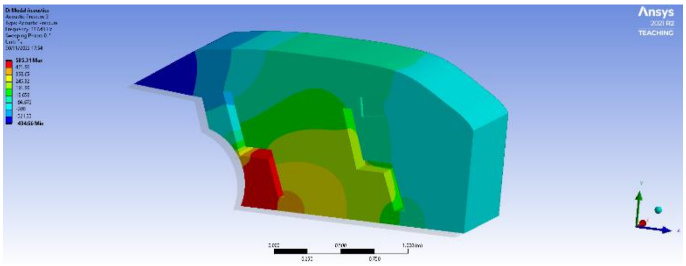

Figure 3 Mode 2 (117.43Hz)

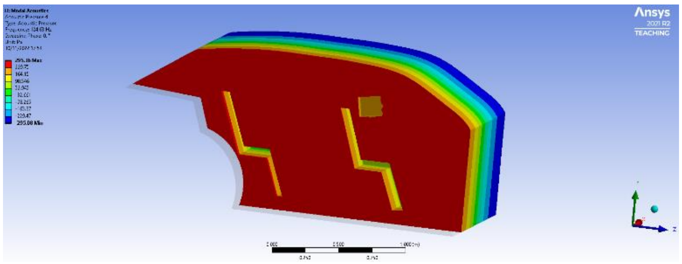

Figure 4 Mode 3 (124.63Hz)

For mode 1, we can see the sound pressure distribution along the vehicle's longitudinal direction; the SPL is highest towards the rear side, around 426Pa. A maximum SPL around the head location of the back passenger can be seen in the sectional view. In comparison to mode 1, the SPL at the rear passenger's head position is reduced in mode 2, and we see movement along the y and z axis, as well as a mid-range sound level throughout the passenger's seating region in the cabin. With reference to the animation, a lateral movement of the sound pressure was observed in the third mode, with the highest pressure level seen towards the side and a relatively lower pressure level at the centre.

Acoustic Harmonic Analysis

In statement to the standard for interior noise measurement, a point around the driver's right ear was selected as a reference to measure the response. This response is used to measure the sound pressure level near the driver's head position. An input source was provided using the surface velocity option. While the absorption coefficient of 0.4 (considering a leather cover on the seat) was specified for the seat surfaces using the absorption surface option.

The frequency response curve for both scenarios is shown in the figures below.

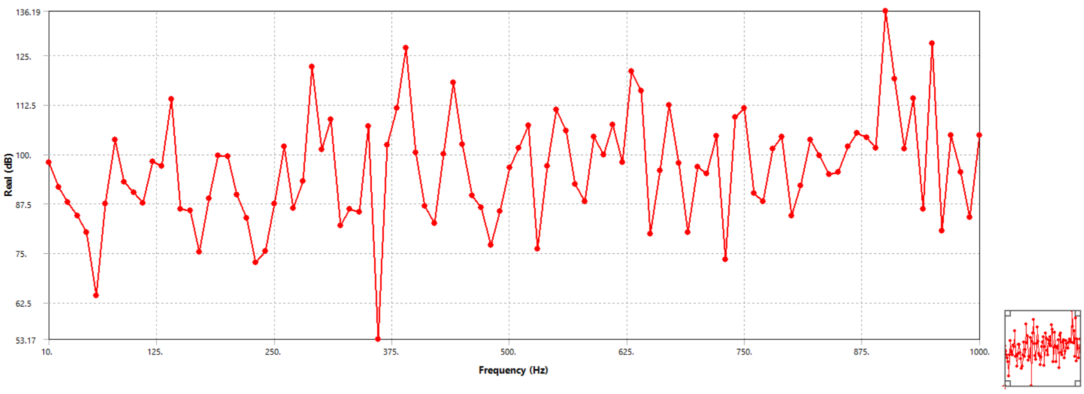

Figure 5 Frequency Response function curve for without absorber condition

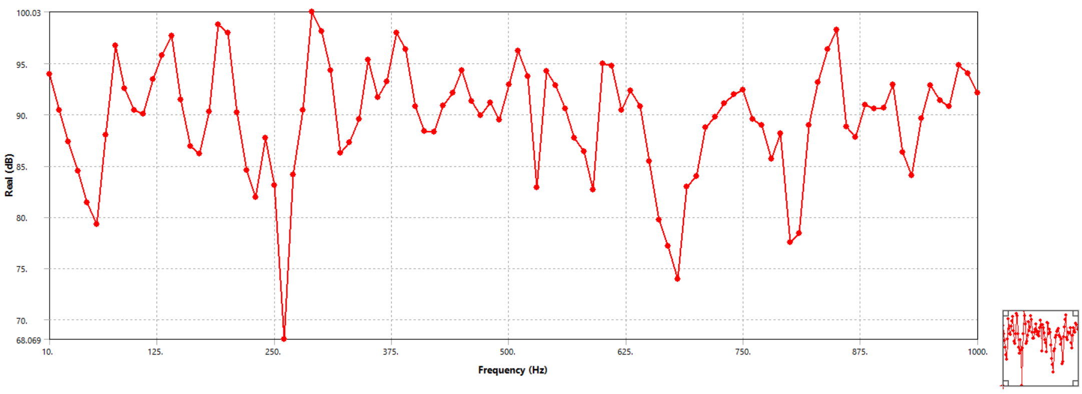

Figure 6 Frequency Response function curve for with Absorber condition

In Figures 5 & 6, it can be observed that the peak amplitudes are at the same frequency range as the modes obtained from the modal analysis. This is because of acoustic resonance which amplifies the input excitation at a frequency near the natural frequencies.

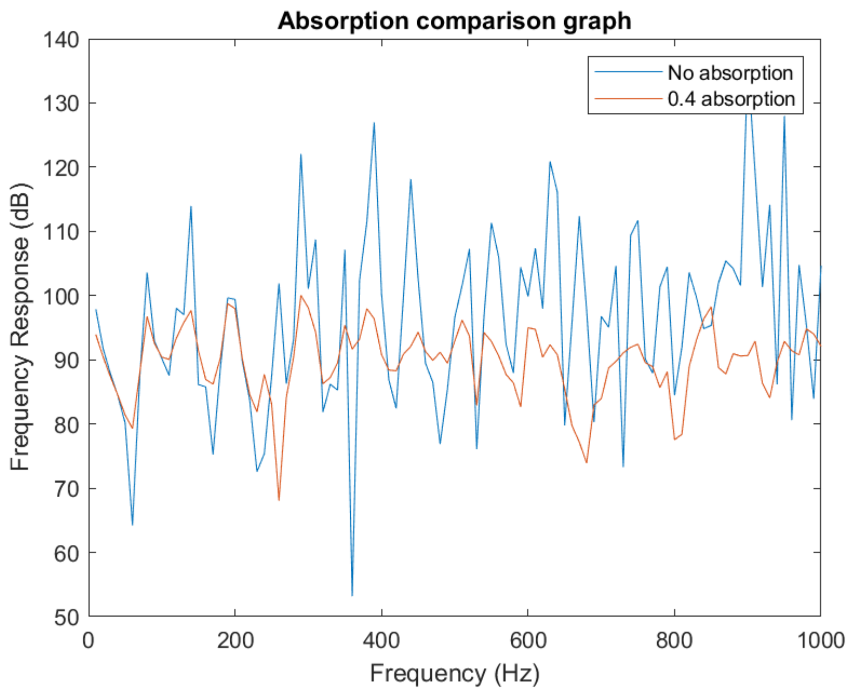

Figure 7 Comparison of Frequency response graph for with and without absorption condition

From the above comparison graph, we can see a reduction in sound level with the consideration of absorbing material co-efficient. This indicates the significant impact of the interior objects on sound absorption which is directly dependent on the surface material and its absorbing capability.

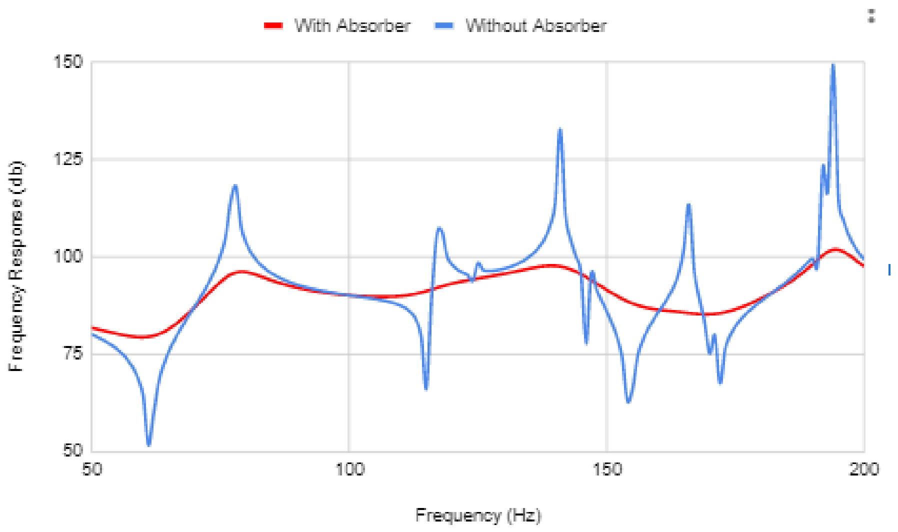

Figure 8 Frequency response curve comparison

The above figure shows the frequency response for the interested range up to 200Hz with an interval of 200 points to obtain better resolution.

This study can be enhanced by considering more accurate external source factors and taking into account all other absorbing coefficients such as the floor carpet, dashboard, roof liner, and others.

Sound source implementation

In this study, we have considered a case of noise generated by engine vibration. The velocity component of a vibrating engine acting normally on the selected surface of the cavity was defined by using the surface velocity option. The following equation was used to calculate surface velocity.

$P_{ref} = 20 * 10^{-6}$ Pa

$P_{rms} = P_{ref} * 10^{\frac{SPL}{20}}$

$V_{rms} = \frac{P_{rms}}{\rho * c}$

where,

$\rho = 1.225$ & $c = 346.25$

In the harmonic analysis, we can also specify a different type of source load on the system such as a mass source.

Reflection

Following the coursework, we learned how effective the finite element method is in predicting the frequency and mode forms of the acoustic system, which affects the noise level inside the cabin, and how this data can help a vehicle development engineer plan for distribution over these frequencies. Furthermore, we observed a difference in sound pressure level with and without the consideration of absorption surface, indicating the influence of seats (in our case) and other objects on the interior noise level. Furthermore, we learned about the parameters needed to conduct a thorough analysis while accounting for precise and additional parameters.

References

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Appendix

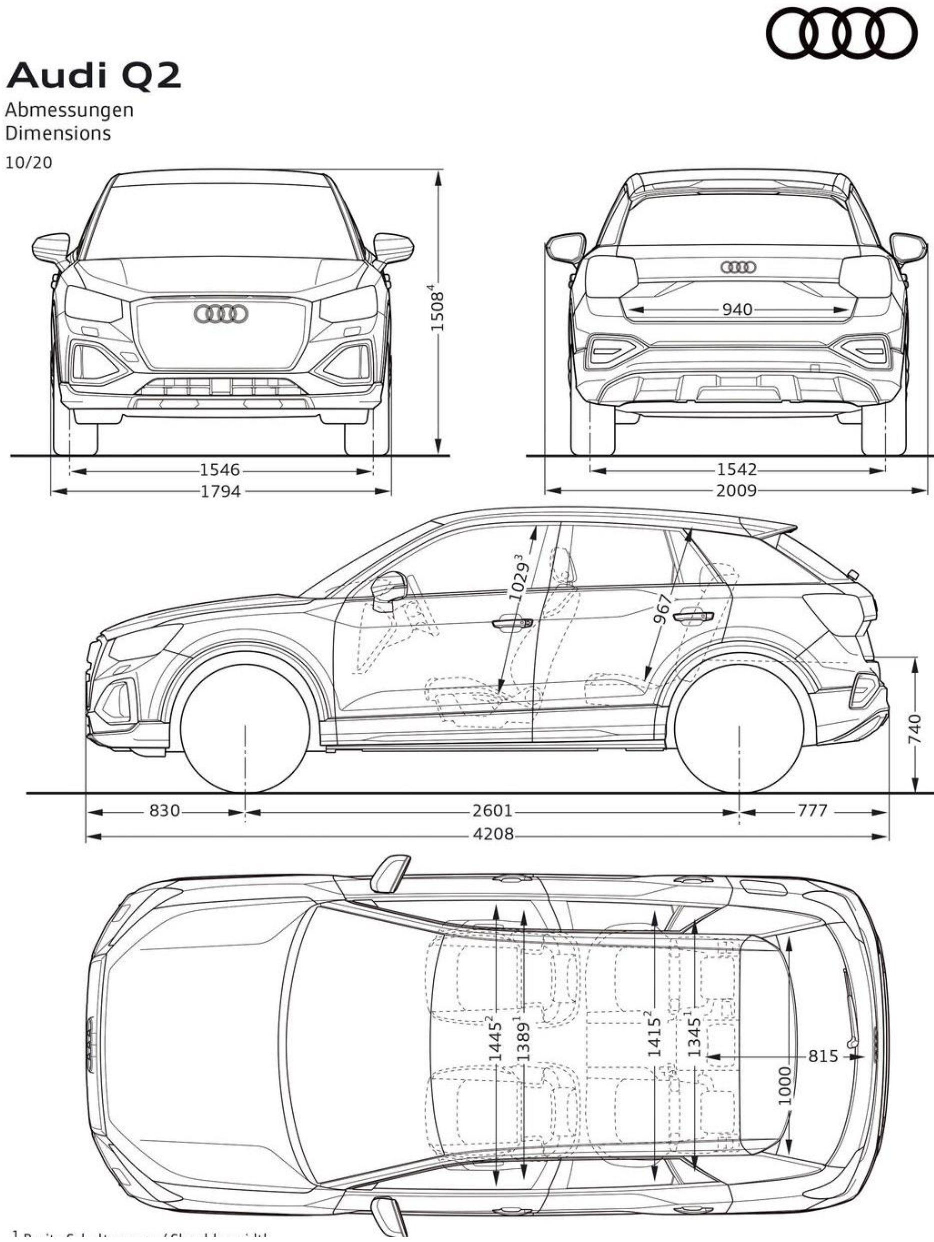

Figure 9 Dimensions of the reference model